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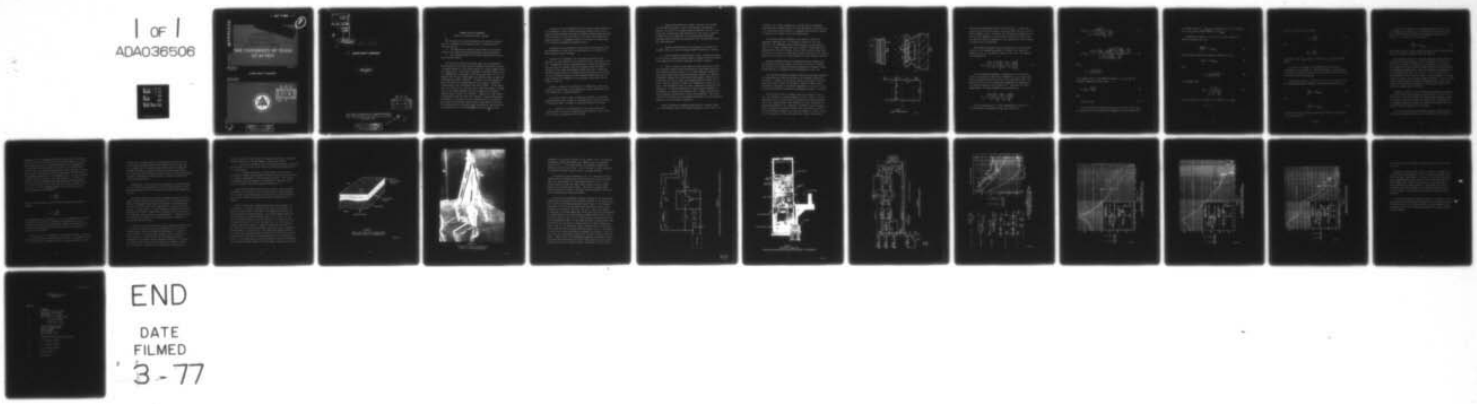

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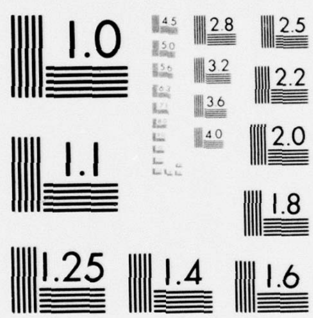

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Herbert V. Hillery
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(6) INTENSE SOUND IN SUBMARINES

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INTENSE SOUND IN SUBMARINES

Herbert V. Hillery and John A. Behrens

The intensity of the sound produced in air inside an interior space of a submarine by an exterior source is controlled by two principal factors:

1. the absorption of airborne sound energy by the interior surface of the walls, by interior furnishings, and by personnel,

2. the transmissibility of the walls separating the interior space from the source.

The sound field within an enclosed space for all frequencies having wavelengths less than twice the greatest horizontal, vertical, or diagonal dimension of the space consists of a highly complex system of standing waves. Since half a wavelength of airborne sound even at the comparatively low frequency of 100 Hz is only about 5.5 ft, it is apparent that for most sounds almost any habitable space is many half wavelengths in greatest dimension. If there is no sound absorption within the space and no sound leaks out of the space, a system of standing waves, once established, will exist forever. If the absorption is not zero, but is very low, only a very small amount of energy, equal to the energy absorbed, needs to be added to the system of standing waves to sustain its intensity. Adding even a small amount of additional energy can cause the intensity to increase drastically. Consequently, the existence of high intensity sound in an enclosed space does not necessarily indicate that a large amount of energy is being transmitted into the space. The high intensity may result from low energy input if the sound absorption within the space is low.

If almost no sound is absorbed inside the space at some frequencies, the transmission of very little sound energy through a wall into the space at those frequencies is likely to produce extremely high intensity within the space. On the other hand, if the walls and furnishings are highly absorptive, the maximum intensity in the space is nearly equal to the intensity transmitted through the walls.

Consequently, the absorption of sound within a space must be high and transmission of sound into the space must be low to achieve low sound intensity within the space.

Reducing the transmission into the space may produce little reduction in sound intensity if the absorption within the space is very low. Furthermore, interior coatings applied to reduce the transmission into the space can actually cause the intensity within the space to increase if they reduce the absorption within the space. Conversely, interior coatings applied to increase sound absorption within the space can cause increased intensity if they increase transmission into the space by making the impedance of the wall more closely match the impedance of the air.

However, generally, the probability of inadvertently increasing intensity is much greater from attempting to reduce transmission than from attempting to increase absorption.

The correct steps to take to reduce the intensity of externally generated sound within a space on a submarine can be listed as follows in descending order of probable dB intensity reduction per dollar.

1. Increase the sound absorption of the walls, floor, and overhead within the space by adding absorptive materials that at least do not increase transmission into the space.

2. Suspend three-dimensional shapes, covered on all surfaces with absorptive materials, in any unused volumes, particularly in corners. All sides of the shapes should be exposed to the air. Such shapes have exceptionally high absorption per unit volume. Addition of shapes may not be practical, but providing something roughly equivalent may be possible by covering all sides of piping and wiring with sound absorbers.

3. Increase the absorption of the surfaces of structures and equipment within the space by adding absorptive materials in patches.

4. Reduce parallelism between walls, floor, overhead, equipment, and structures by adding prismatic shapes or shapes with curved surfaces before applying the absorptive materials of Steps 1 through 3.

5. Steps 1 through 4 should make the space acoustically "dead." Even partially followed, they should produce a significant reduction in sound intensity within the space. If a trial temporary installation, which can be made without installing permanent fasteners or otherwise permanently altering the interior walls and equipment, shows that the sound level is still too high, the absorbing materials called for in Step 1 should be modified to reduce the transmission of sound into the space. The modification must not cause a significant reduction in the absorption. If reduced transmission of only a single narrowband of frequencies is necessary, the reduction can be readily achieved with a mass-spring-mass coating tuned to reject the narrowband. The exposed surface of the coating can be perforated and backed with sound absorbing material to provide acoustic absorption.

Before considering in somewhat greater detail a possible design for a satisfactory coating for reducing transmission into a space on a

submarine, let us first consider how a typical anechoic underwater sound acoustical coating, useful for other applications, would function if it were used on the interior of a submarine, and why such a coating is unlikely to be useful in the latter application.

The typical anechoic underwater sound coating is made of an elastomeric material, and it is designed to reflect as little as possible of the underwater sound of some frequency ω_0 that impinges upon it. In order to explain why such a coating is inappropriate for application to the inner surface of a hull wall to reduce sound transmission into the hull, it is desirable to first consider how the coating, applied to the exterior of a hull, performs its echo reduction function.

The echo reduction process is perhaps most easily understood by considering an electrical analog circuit for the coating. Therefore let us derive the classical electrical analog for an elastomer coating placed exterior to a submarine hull as represented in Fig. 1(a).

The classical analog relationships between corresponding parameters for acoustical and electrical systems for the impedance analogy are as follows: Voltage in an electrical circuit is analogous to acoustic pressure, electric current is analogous to acoustic volume current, inductance is analogous to inertance, capacitance is analogous to compliance, and electrical resistance is analogous to acoustic resistance.

The steel hull with its coating, shown in an enlarged view in Fig. 1(b), behaves as a large inertance L_s for the coating. Acting in combination with the voids inside the elastomer, the hull mass allows very little coupling between the inertance L_α of the coating and the terminating impedance, Z^*_0. Therefore, the motion of the inertance L_α, which is formed by the effective mass of the void closure, is driven by the sound field through the two parallel elements, C_α and R_α, where C_α is the compliance of the void walls and R_α results from viscous

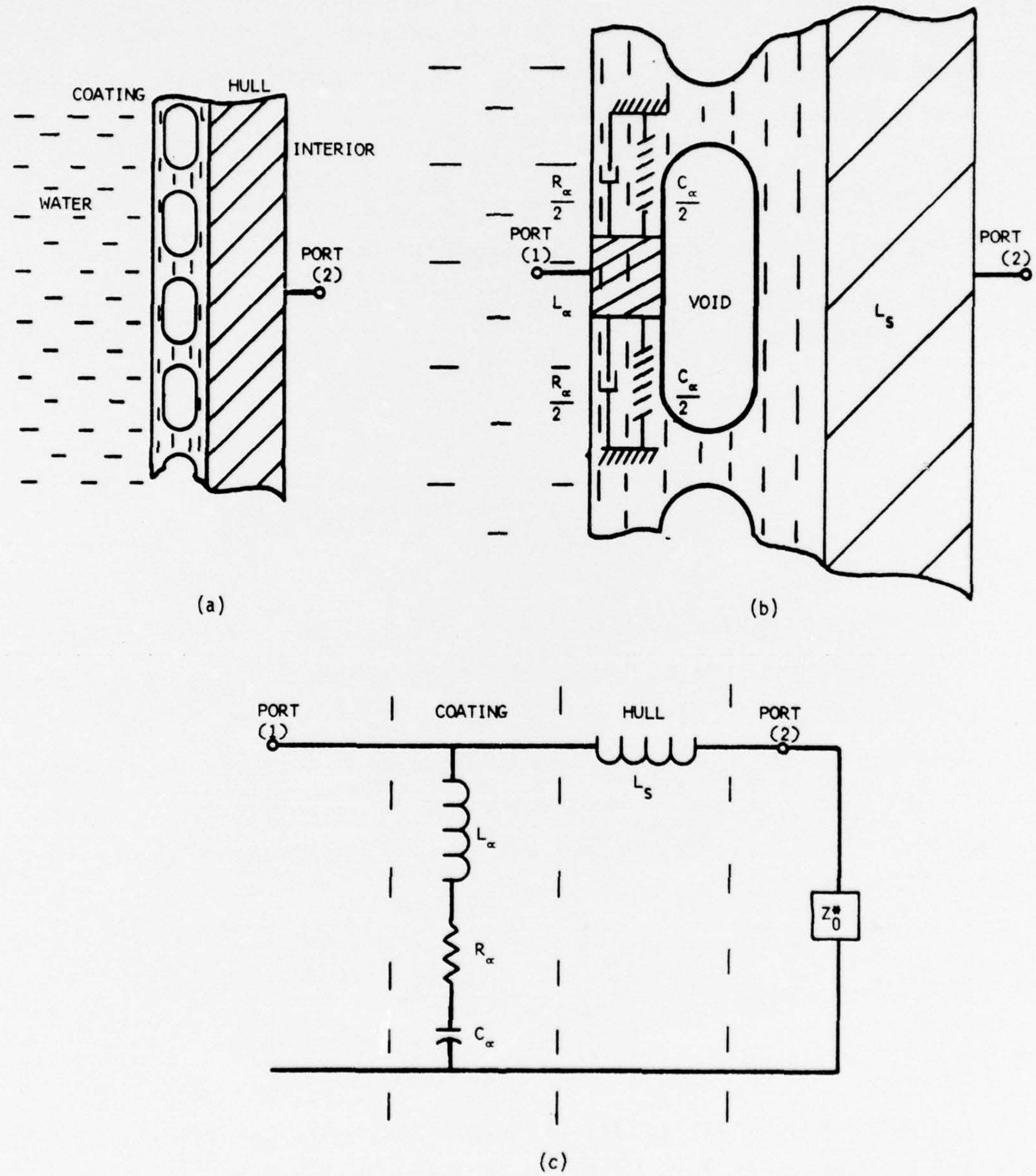

FIGURE 1
EXTERIOR COATING AND MODEL

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shear losses accompanying motion of the void walls. C_α and R_α are coupled to ground, or the inertial reference, which lies at a position between adjacent voids. Of course, for the previous statements to be true the bulk compliance of the elastomer without voids must be much less than the effective compliance of the combination of void closures, void walls, and bulk elastomer.

The classical equivalent circuit representation of the coating, steel hull, and terminating impedance in terms of the acoustical elements L_α, C_α, R_α, L_s and Z^*_0 is shown in Fig. 1(c). The complex input impedance is

$$Z^*_{in} = \frac{\left(j\omega L_s + Z^*_0\right)\left\{R_\alpha + j\left(\omega L - \frac{1}{\omega C_\alpha}\right)\right\}}{R_\alpha + Z^*_0 + j\left\{\omega(L_s + L_\alpha) - \frac{1}{\omega C_\alpha}\right\}} \quad . \tag{1}$$

If the terminating impedance is assumed to be a semi-infinite volume of air having a characteristic impedance of $\rho_a c_a$, and a plane wave is radiated from the hull into the air, then Z^*_0 is real. The magnitude of Z^*_0 for air is small when compared to the magnitude of ωL_s and when compared to most values of R_α. Therefore to a close approximation, the input impedance can be rewritten as

$$Z^*_{in} = \frac{\left(j\omega L_s\right)\left\{R_\alpha + j\left(\omega L_\alpha - \frac{1}{\omega C_\alpha}\right)\right\}}{R_\alpha + j\ \omega\left(L_\alpha + L_s\right) - \frac{1}{\omega C_\alpha}} \quad . \tag{2}$$

After rationalizing Eq. (2) and simplifying the results, the real and imaginary components of Z^*_{in} can be written as

$$\text{Re } Z^*_{in} = \frac{\left(\omega L_s\right)^2 R}{R_\alpha^2 + \dfrac{(L_s + L_\alpha)}{C_\alpha}\left(\dfrac{\omega}{\omega_0} - \dfrac{\omega_0}{\omega}\right)^2} \quad , \tag{3}$$

and

$$\text{Im } Z^*_{in} = \frac{\omega L_s R_\alpha^2 + \omega L_\alpha\left(\omega L_\alpha - \dfrac{1}{\omega C_\alpha}\right)\left(\sqrt{\dfrac{(L_s + L_\alpha)}{C_\alpha}}\right)\left(\dfrac{\omega}{\omega_0} - \dfrac{\omega_0}{\omega}\right)}{R_\alpha^2 + \left(\dfrac{L_s + L_\alpha}{C_\alpha}\right)\left(\dfrac{\omega}{\omega_0} - \dfrac{\omega_0}{\omega}\right)^2} \quad , \tag{4}$$

where

$$\omega_0 = \sqrt{\frac{1}{C_\alpha(L_s + L_\alpha)}} \quad .$$

It is apparent that at the angular frequency $\omega = \omega_0$, the real and imaginary components of Z^*_{in} become

$$\text{Re } Z^*_{in} = \frac{(\omega_0 L_s)^2}{R_\alpha} \quad , \tag{5}$$

and

$$\text{Im } Z^*_{in} = \omega_0 L_s \quad . \tag{6}$$

For the coating to be nearly anechoic the real part of Z^*_{in} must be nearly equal to the characteristic impedance of water per unit area

of incident sound, $Z_0 = (\rho c)_{water}$, and the magnitude of the imaginary part of Z^*_{in} must be small relative to $(\rho c)_{water}$.

Consequently, from Eqs. (5) and (6), the first condition for large echo reduction at ω_0 is

$$\frac{(\omega_0 L_s)^2}{R_\alpha} \simeq (\rho c)_{water} \quad , \tag{7}$$

and the second condition for large echo reduction at ω_0 is

$$|\omega_0 L_s| << (\rho c)_{water} \quad . \tag{8}$$

Since

$$\omega_0 = \sqrt{\frac{1}{(L_s + L_\alpha)C_\alpha}} \quad , \tag{9}$$

it is apparent that

$$\omega_0 L_s = \sqrt{\frac{L_s}{C_\alpha\left(1 + \frac{L_\alpha}{L_s}\right)}} \quad . \tag{10}$$

Since the steel hull is likely to be at least 0.5 in. thick,

$$L_s >> L_\alpha$$

and Eqs. (9) and (10) can be written

$$\omega_0 \simeq \sqrt{\frac{1}{L_s C_\alpha}} \tag{11}$$

and

$$\omega_0 L_s \simeq \sqrt{\frac{L_s}{C_\alpha}} \quad . \tag{12}$$

Therefore, to make $|\omega_0 L_s|$ small, the ratio of L_s to C_α must be made small.

If ω_0 and L_s are given, as is generally the case, only the compliance C_α, the inertance L_α, and the resistance R_α can be altered to satisfy the conditions of Eqs. (7) and (8), which must be satisfied to achieve a large echo reduction.

Substituting from Eqs. (11) and (12) into Eqs. (7) and (8) yields the following approximate conditions for high echo reduction at ω_0:

$$\frac{L_s}{C_\alpha R_\alpha} = (\rho c)_{water} \tag{13}$$

and

$$\sqrt{\frac{L_s}{C_\alpha}} << (\rho c)_{water} \quad . \tag{14}$$

Equation (13) can be written in terms of the effective loss factor η for the coating, where

$$\eta = \omega_0 R_\alpha C_\alpha \quad . \tag{15}$$

The value of η normally lies somewhere between the value of the bulk modulus loss factor $\eta_B(\omega, t)$ of a material, which can be as high as 0.1, and the shear loss factor of the material, which can be as high as 5. In terms of η, Eq. (13) becomes

$$\frac{\omega_0 L_s}{\eta} = (\rho c)_{water} \quad , \tag{16}$$

which gives some idea of the overall loss factor that a coating material must have in order to provide a high echo reduction.

The acoustical behavior of a properly designed exterior anechoic coating at the angular frequency ω_0 can be explained as follows: As can be seen from Figs. 1(b) and 1(c), the inertance L_s of the steel hull tends to block the acoustical volume velocity from flowing through the terminating impedance Z^*_0 so that little pressure is developed at the terminating port, port (2). Since $L_\alpha << L_s$, the effect of L_α on the performance of the circuit is small. If the compliance C_α is large enough, C_α tends to short the input signal to ground. At ω_0, the volume velocity flows through the effective resistance $L_s/C_\alpha R_\alpha$ which nearly matches the characteristic impedance of the water and little of the incident acoustic energy is reflected.

Ideally, to achieve the best acoustical performance a very large C_α would be chosen for the coating as shown by Eqs. (13) and (14), but because the coating must perform its function under hydrostatic loads, the largest C_α that can be chosen is limited to the C_α permitted by the maximum hydrostatic pressure the coating must withstand and by the maximum thickness permitted for the coating.

Let us now consider how an exterior submarine coating designed for high echo reduction would act if it were placed on the inside of a

submarine hull. The analogous electrical circuit would be similar to the circuit of Fig. 1(c) except that the shunt elements C_α, R_α, and L_α would be placed at port (2) in parallel with the low impedance Z^*_0. If the interior of the submarine is assumed to be a semi-infinite volume of air, then Z^*_0 can be assumed to be real and equal to $(\rho c)_{air}$. Now, in order to prevent the acoustic signals that come into port (1) from appearing across Z^*_0, it is necessary that L_α, R_α, and C_α provide a "short circuit" across Z^*_0. The only way that L_α, R_α, and C_α can provide a very low impedance shunt path around Z^*_0 at a frequency ω_0 is to have the value of R_α be much less than $(\rho c)_{air}$ and also either to have $L_\alpha \approx 0$ and C_α extremely large or to have L_α and C_α act in series resonance at the frequency

$$\omega_0 = \sqrt{\frac{1}{L_\alpha C_\alpha}} \quad .$$

Since it was shown above that the exterior coating must satisfy the equation

$$\omega_0 \approx \sqrt{\frac{1}{L_s C_\alpha}}$$

to be anechoic at ω_0, and since $L_s >> L_\alpha$, it is apparent that a coating designed for exterior use is unlikely to be useful inside a hull. Virtually all of the signal that passes through the hull wall will be applied to Z^*_0 whether or not the inner surface of the hull wall is covered with the coating.

However, despite the validity of the previous argument, one cannot say for certain that the coating would be useless. The argument assumes that the sound passes through the wall as a longitudinal wave and that

the wall acts as a rigid piston in radiating the sound into the air. If the sound transmission results from flexural motions of the wall, it is possible that the coating could fortuitously act to damp the flexural vibrations and reduce transmission; or the coating might, if its front and back surfaces are weakly coupled in shear, fortuitously interpose a transmission reducing acoustical resistance between the flexing hull and the air.

Nevertheless, it can at least be concluded that a coating designed for use in water on the exterior of a hull is highly unlikely to have optimum characteristics for use inside a hull for reducing transmission into air.

The optimum coating for use inside a submarine might well have an analog circuit similar to the analog circuit for the exterior coating. However, for maximum effectiveness, the interior coating should have a value of C_α that is many times larger than the C_α for the exterior coating. A large value of C_α can be obtained by changing the material from which the coating is made from an elastomer with a few voids to a more compliant material, such as a compliant foam. A small mass would be attached to the surface of the foam to provide an inertance L' between the foam and the air.

In order to check the thesis that a high compliance coating would be effective for reducing the sound transmission into an enclosed submarine space, ARL has performed some experiments on a couple of coating samples. Only small coating samples could be readily constructed and tested, and therefore the relationship between the performance of the samples and the probable performance of the large area interior coating that would be needed in a submarine cannot be precisely defined. However, if the following two assumptions are

valid, the results of testing the small samples should give a reasonably accurate indication of the performance of the actual coating.

1. If a hull is vibrating only in flexural and thickness vibrations, then an area A of the hull can be treated as a vibrating rigid piston if A is small enough.

2. In a large area coating, a small element of area A can be sufficiently isolated from surrounding similar elements to permit element A to operate independently of the surrounding elements. The isolation of individual areas can probably be accomplished by dividing a mass inside the coating into small squares as shown in Fig. 2.

Since neither assumption appears to be untenable, the preliminary measurements ARL has made on the two simple coating elements probably give at least a rough indication of the reduction in sound transmission that could be obtained with coatings similar in construction to the samples.

The two elements were constructed with brass for the masses, as shown in Fig. 2, and with a single layer of cell tight rubber in one case and a double layer in the other. The elements were studied and tested using three different methods. In the first method acoustical tests were made on a sample element in ARL's 3 in. diam, 17 ft long pulse tube shown installed in its well in Fig. 3. The tube was filled with water to within 3 in. of the top where a 1/2 in. steel plate and a sample element were placed. A pulse of sound enters the water at the lower end of the tube and, halfway up the tube, passes a hydrophone, which measures its pressure level. The sound then passes through the steel plate and the sample element and enters the air where an isolated microphone (not shown) picks up the pressure level of the transmitted sound. In the second method, electrical analog circuits were determined for the single and double layer cell tight rubber elements, and voltages

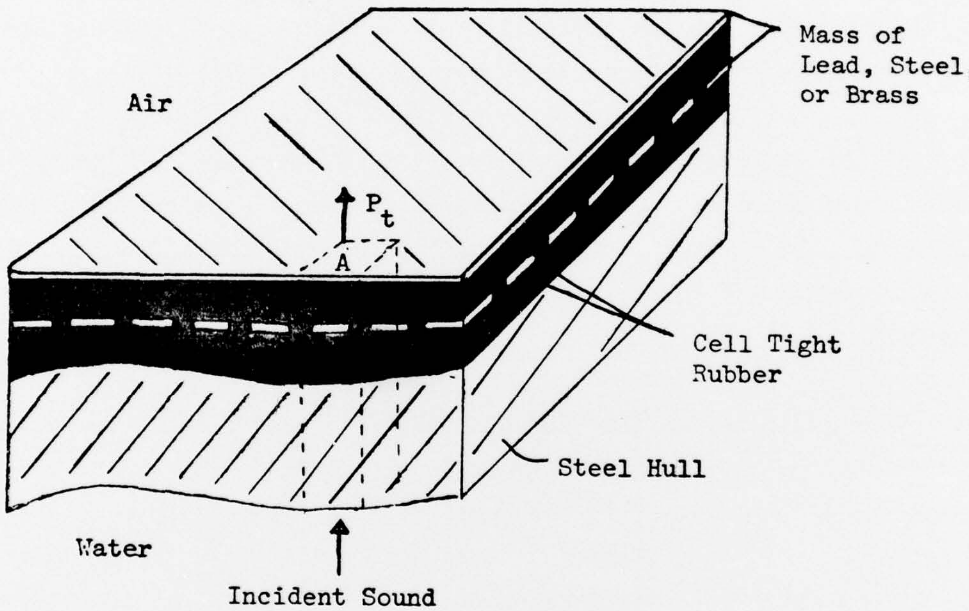

FIGURE 2

DOUBLE LAYER CELL TIGHT RUBBER BAFFLE
MOUNTED ON INSIDE OF PRESSURE HULL

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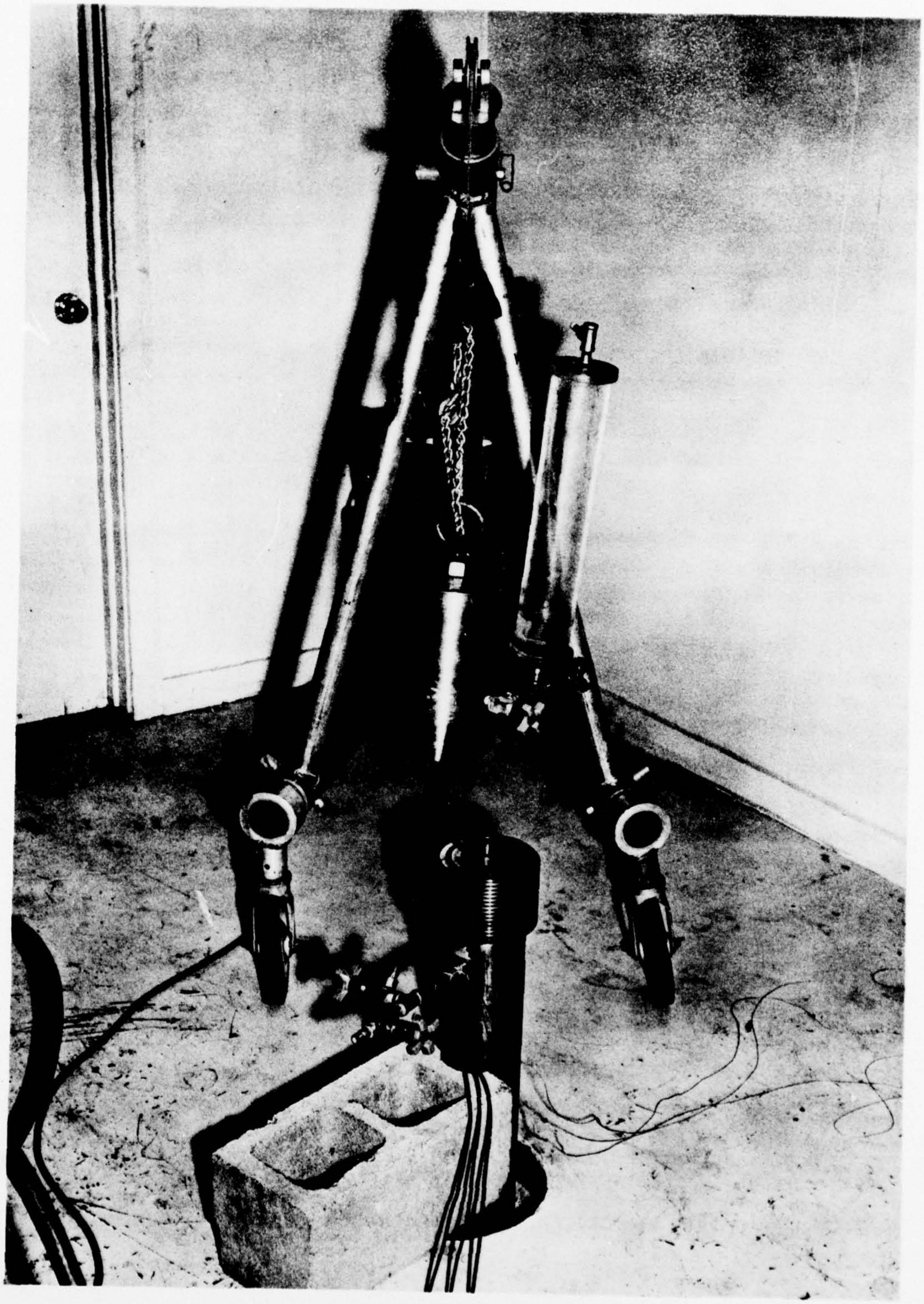

FIGURE 3

THE END CAP READY FOR MOUNTING ON THE 17 ft IMPEDANCE TUBE

analogous to the pressure incident on the elements and to the pressure transmitted through the elements were measured using the equipment shown in Fig. 4. In the third method, the impedance computer and mechanical shaker shown in Figs. 5 and 6 were used to obtain mechanical acceleration transfer functions for the two sample elements.

The acoustical results measured in the 3 in. diam pulse tube are compared with the analog circuit predictions for the single and double layer cell tight rubber 3 in. diam baffle elements in Fig. 7. The analog results show that the double layer element should be greatly superior to the single layer element. The reason the double layer element performed so poorly in the pulse tube is believed to have been due to flanking paths around the element.

The mechanical acceleration transfer function for the single layer element and the analogous current transfer function of its analog circuit were plotted versus frequency and are shown in Fig. 8. The analogous capacitance of 4.3 μf was determined from the measured static spring stiffness of the element, 190 lb/in. per sq in. The 0.03 H inductor is analogous to the approximate mass of the brass plate and accelerometer. The sudden upswing in the acceleration transfer function curve in Fig. 8 is due to flexing of the 0.025 in. brass top mass caused by the inertia of the accelerometer mass. It was found that when a thicker brass plate was used for the top mass, the flexing moved to a higher frequency as shown in Fig. 9. The results of using a different type of foam along with a lower mass for the output plate are shown in Fig. 9. By changing the mass of the output plate and the thickness or stiffness of the cell tight foam, the antiresonant frequency can be moved to any desired frequency. Figure 10 gives the mechanically measured acceleration transfer function of the double layer cell tight rubber element and the current transfer function of its analog circuit. Flexure of the output plate

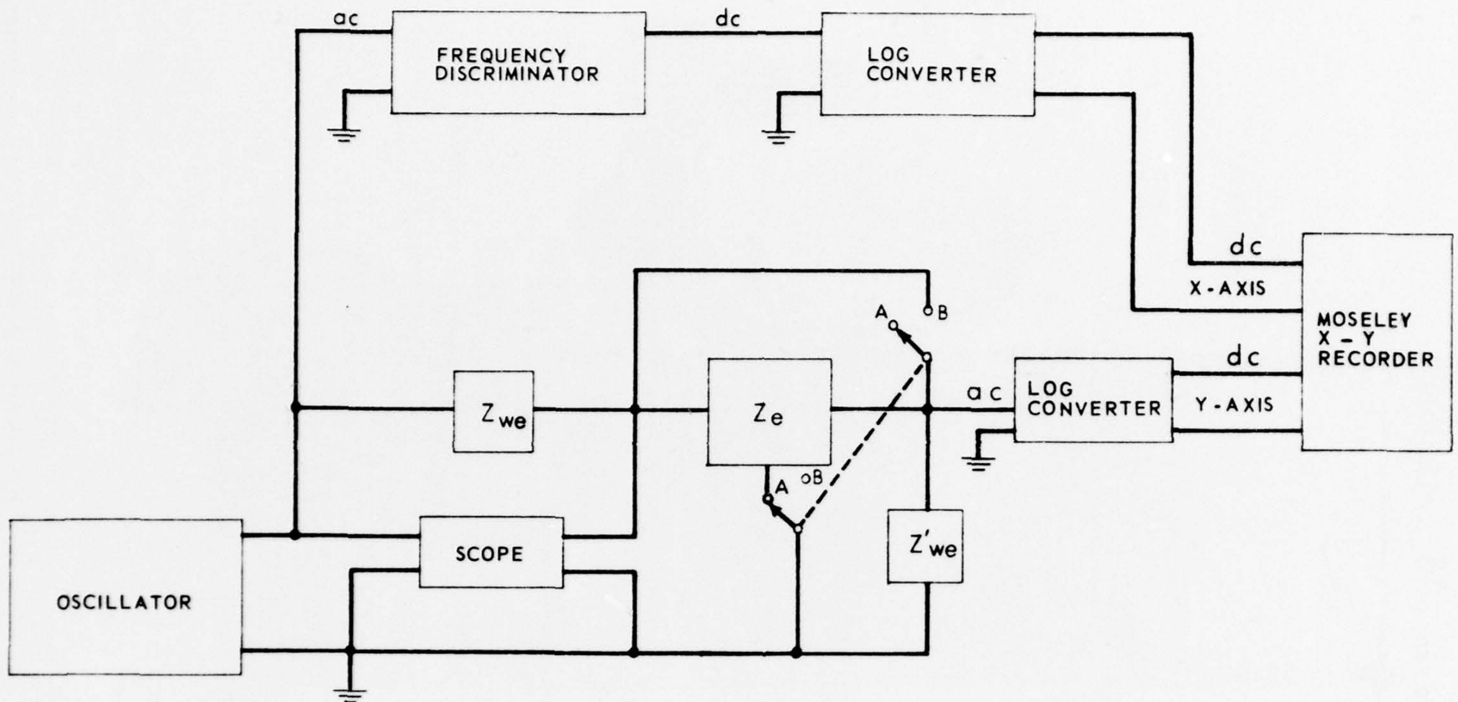

FIGURE 4
BLOCK DIAGRAM OF TEST CIRCUIT FOR PLOTTING TRANSMITTED INTENSITY $\overline{I}_T$

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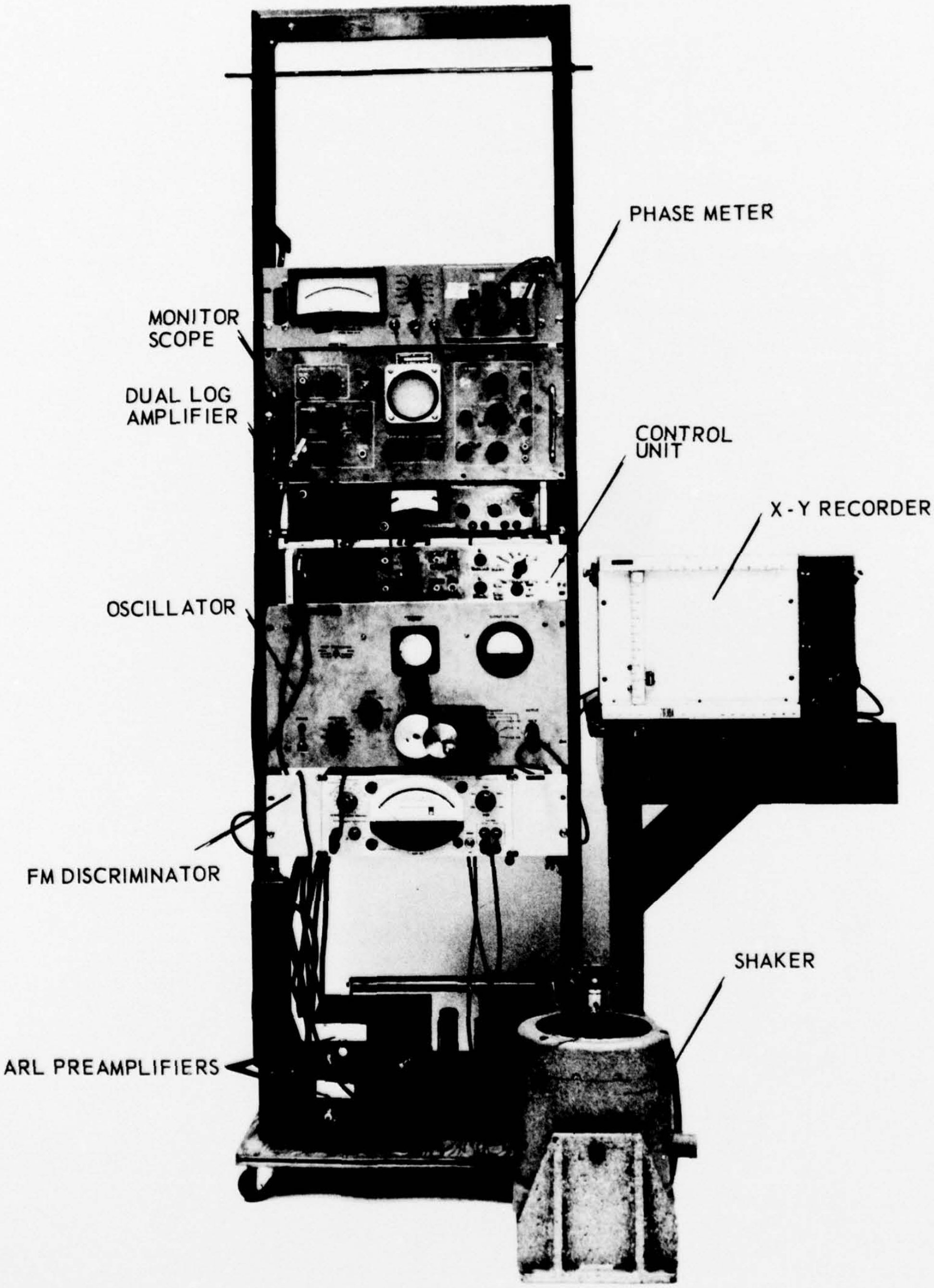

FIGURE 5
IMPEDANCE COMPUTER
SETUP FOR RECORDING MECHANICAL IMPEDANCE vs FREQUENCY

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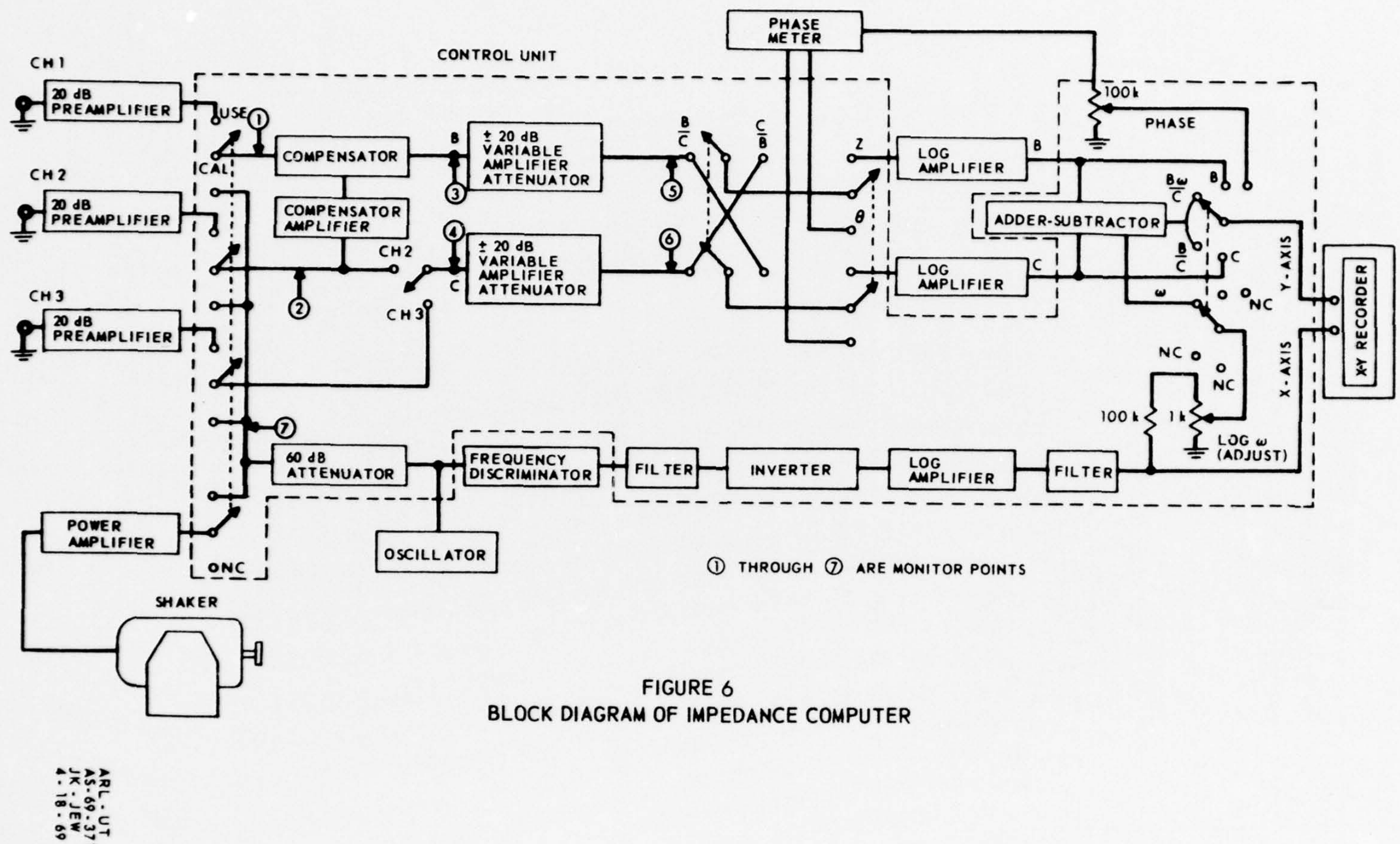

FIGURE 6
BLOCK DIAGRAM OF IMPEDANCE COMPUTER

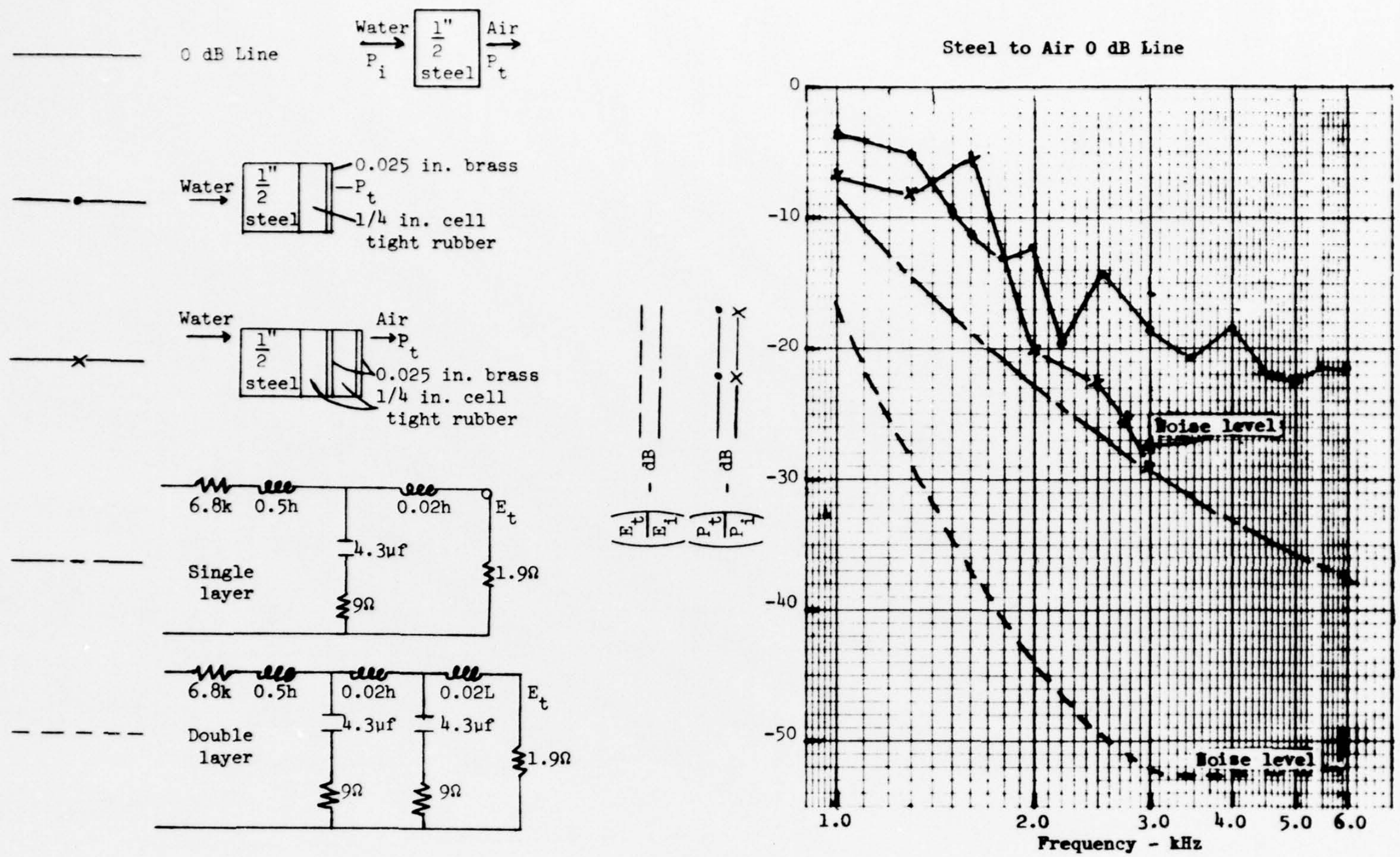

FIGURE 7
ACOUSTICAL AND ANALOG CIRCUIT RESULTS
FOR SINGLE AND DOUBLE LAYER CELL TIGHT RUBBER
3 INCH DIAMETER BAFFLE ELEMENTS

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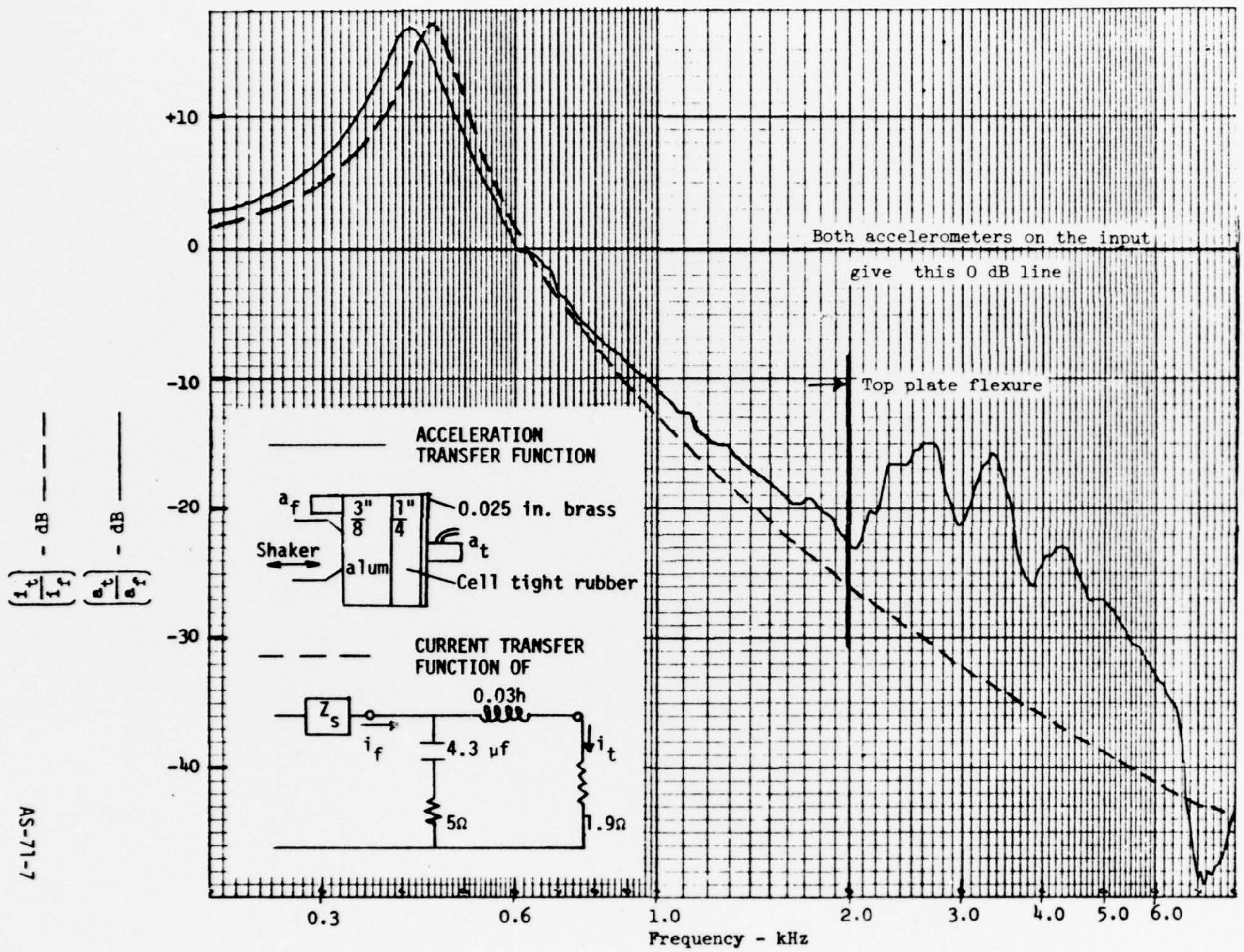

FIGURE 8
MECHANICAL AND ANALOG CIRCUIT RESULTS
FOR A SINGLE LAYER CELL TIGHT RUBBER BAFFLE ELEMENT

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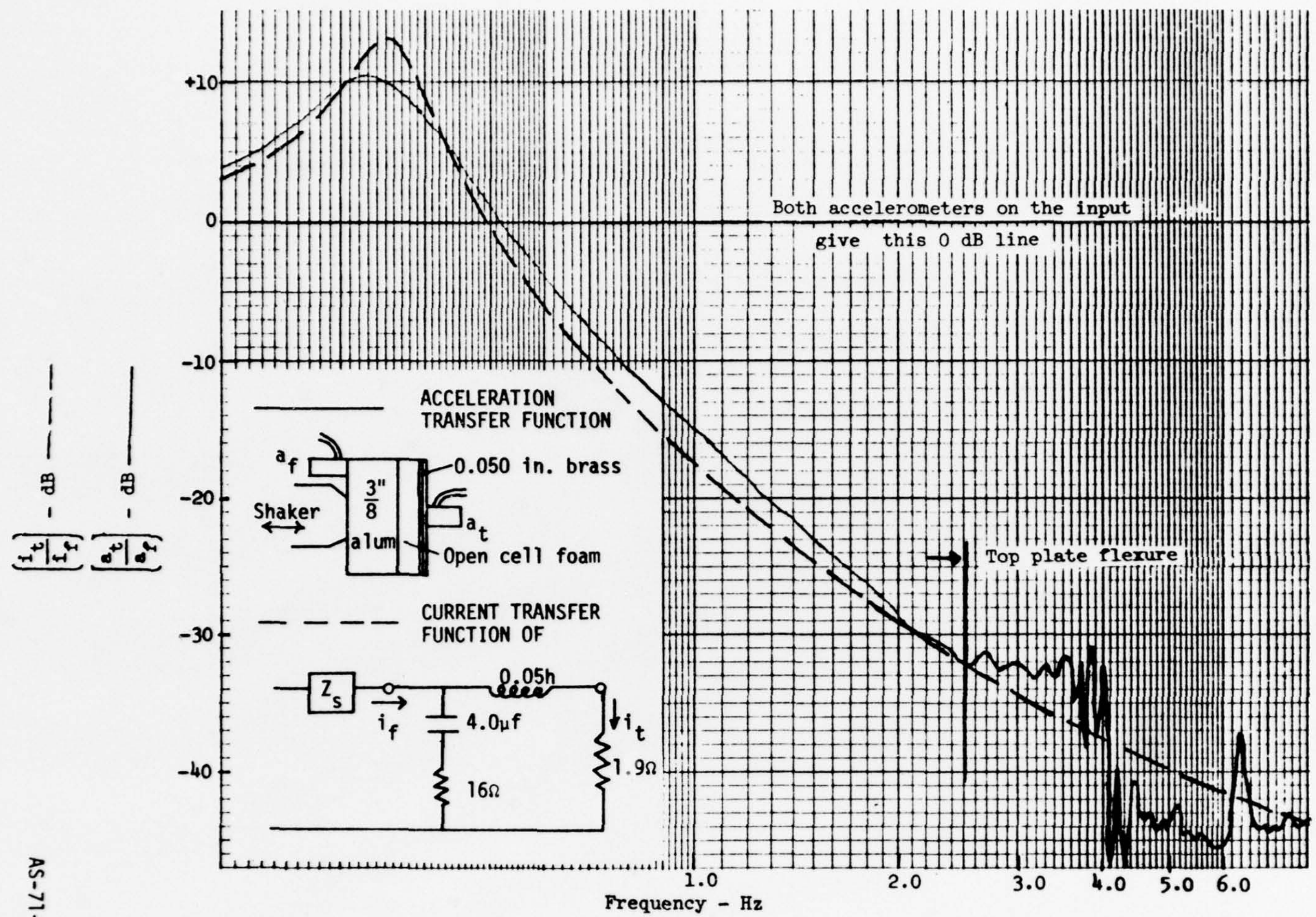

FIGURE 9

MECHANICAL AND ANALOG CIRCUIT RESULTS
FOR A SINGLE LAYER OPEN CELL FOAM BAFFLE ELEMENT

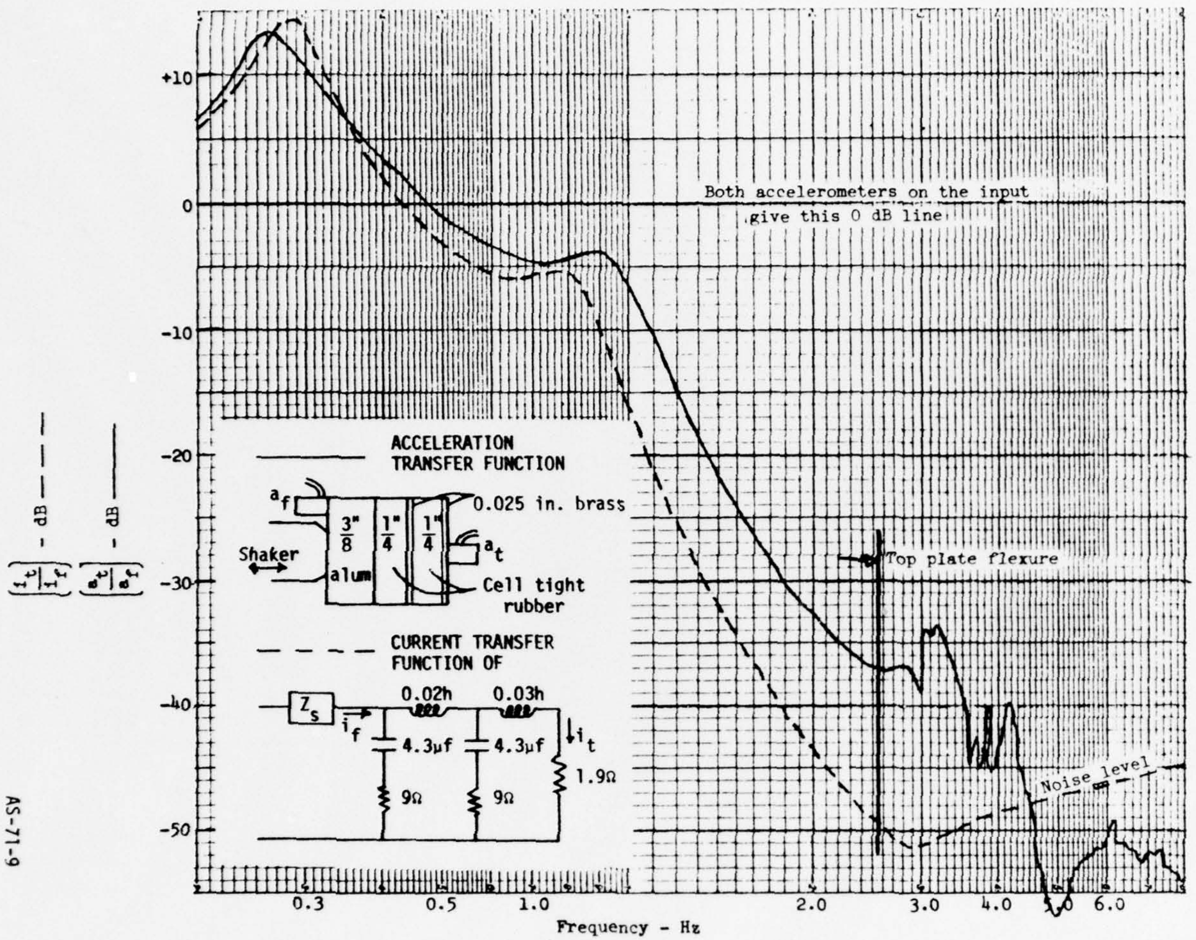

FIGURE 10

MECHANICAL AND ANALOG CIRCUIT RESULTS FOR
A DOUBLE LAYER CELL TIGHT RUBBER BAFFLE ELEMENT

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produced the erratic acceleration transfer between 3 and 6 kHz shown on Fig. 10.

Analysis of the data given in Figs. 7 through 10 indicates that a single layer element made of 1/4 in. thick cell tight rubber and a thin brass plate of mass/area = 3.3 g/sq in. will reduce sound transmission approximately 25 dB at and near 3 kHz. The data show that with a double layer element, 1/2 in. thick, a 40 dB reduction in sound transmission should be achieved. By increasing the mass or decreasing the spring stiffness in the elements, greater isolation can be obtained. Also by using more layers, more isolation can be obtained.

The type of foam and the amount of mass to be used have not been optimized in these preliminary experiments, but even these perfunctory tests suggest that a very large reduction in sound transmission into submarine interiors can be readily achieved with a properly designed interior coating.

6 January 1971

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